TOWNE
378.748
POS 1899.8

UNIVERSITY of
PENNSYLVANIA
LIBRARIES

SEAS LIBRARY

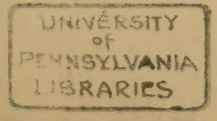
Please return this book as soon as
finished with it. It
returned

2537
990
(slip)

ANALYSIS
OF A
COMPOUND
BELT DRIVEN
AIR COMPRESSOR.

Wm. C. Biddle Jr.
H. W. Reynolds.

Biddle & Reynolds '99.
Air Compressor.
A
89



TOWNE
378.748
P OS 1899.8

UNIVERSITY
of
PENNSYLVANIA
LIBRARIES

4 June '98. Sec. E. Univ. ... B. W. Canby, Jr.

The object of this thesis was to determine the efficiency of compression, the volumetric efficiency the Horse Power required to compress 1000 cu. ft. of air, and to draw the combined cards.

Description of Machine.

The machine was built by Pedrick and Ayer of Phila. and is a compound belt-driven one. It is situated in the Mechanical Laboratory of the University of Pennsylvania and is used for experimental work.

Regarding it as unnecessary to describe the machine in detail again, we refer to a thesis on the same machine, which contains full details, in '98 thesis by Vincent.

The machine was first thoroughly overhauled and put into seeming good condition but considerable difficulty was experienced in the matter of the bearings heating up, making a long run impossible.

Thirty minutes was found to be the longest run obtainable, before the bearings heated up to a dangerous point.

The bearings were examined and looked to be in good condition, so we were unable to decide the cause of this heating.

No one bearing in particular but all heated up.

The brasses heated up very quickly.

We then proceeded to make an indicator rig and to get the clearance. We decided to run an incorrect rig and afterwards correct the cards, because we thought that to make a correct rig would take up too much time. So we turned up a 2" crank, heavy enough not to give, and fastened it to the end of the shaft, and from this we took the motion.

Clearance Volume.

To get the clearance volume we blocked up one end of the machine and took off the box beneath the machine containing the clearance spaces.

We then determined the clearance

volume for the two cylinders in the following manner:

The clearance for the Low Pressure cylinder is made up of the volume of the fin inlet valve openings, plus the volume of the two low pressure discharge valves openings, plus the volume below the low pressure piston to valve openings when the piston is at the bottom of its stroke.

The fin valves being of irregular form were filled with water, the weight taken and reduced to volume at the temperature of the water used. The volume was found to be 3.80 cu. in.

The two valve openings were of regular form and were accurately measured and their volume

found to be 1.924 cu. in.

The volume below the piston when at the bottom of its stroke was also found by accurate measurement and found to be 3.85 cu. in whence the total low pressure clearance = 5.8 + 1.924 + 3.85 = 11.57 cu. in = .0067 cu. ft.

High Pressure Clearance.

This clearance consists of the volume of four valve openings plus the volume under high pressure piston to valve openings when at the bottom of its stroke.

The former by measurement was found to be 3.299 cu. in. and the latter 1.197 cu. in.; whence the total high pressure clearance = 3.299 + 1.197 = 4.497 cu. in = .0026 cu. ft.

The volume in between the cylinders

also acts as clearance and will be used in further determinations.

Its volume was found by filling with water, weighing and reducing to volume. It was found to be 91.8 cu. in. = .0531 cu. ft.

Methods for measuring the Air used.

To get the amount of air used we cased the inlet valves in a wooden casing, airtight, leaving an opening large enough to hold the small end of a galvanized iron cone in the outer end of which the anemometer was placed.

After a few runs we found from comparing the air used calculated from the Anemometer readings and from the piston displacement that the first came nearly four

the second; an absurd result since the piston displacement gives the largest possible amount.

This was probably due to the air pulsations acting on the vanes, then being such a small area in which the air could enter, the momentum thus produced making the anemometer read high.

So we procured a large wooden box (about 3' x 3' x 3'), placed it over the end of the galvanized iron cone. The anemometer was placed in a hole of known dimensions in the side of the large box, opposite the cone.

This box acts as a reservoir and tends to minimize the effects of the pulsations since they now act on a comparatively large

volume of air.

The result was a decided reduction in the anemometer readings and the draught at the opening was steady. But the amount of air due to this method was still much greater than that obtained by the displacement method, the error being probably due to the incorrect reading of the anemometer although we used the calibration given by the maker of the instrument.

Having no means at hand by which we could accurately calibrate the instrument and no time for making an apparatus we decided to abandon this method and obtain the amount of air used by the displacement method, making corrections.

Now, there are three causes which tend to reduce the volume of air actually taken in by the machine from that given by the actual piston displacement.

(1) Valves do not open at beginning of stroke.

(2) Clearance in low pressure compression

(3) Leakage past piston and through valves.

The first to a certain extent includes the second since the effect of the second is only to decrease the available volume due to displacement by the clearance air expanding to the atmospheric pressure, thus partly filling the cylinder.

But since the valves do not open till the pressure equals that of the atmosphere, or slightly below,

If we take as the actual volume the volume proportional to the distance AC (Heat cards, Diagram I), or when the expansion line crosses the atmosphere line, at this point we are sure that the valve opens although it may close again before it remains open for the completion of the stroke, as shown on the actual cards, Figs I and II, by the line recrossing the atmosphere line

The third cause is small and can be neglected. This is due to the area of the rubbing surface being covered with oil which prevents the air from working past.

This method will therefore give very approximately the amount of air entering. It is at least more

accurate than the method of displacement often used, for the reasons given in the above discussion.

Tests.

Three tests were made, the pressure in the receiver, made of a piece of 10" pipe about 5' long, being kept at approximately 70, 80, and 90 lbs., by means of a cock attached to a waste pipe leading from the receiver

Instruments used.

1. Anemometer - as described above.
2. Gauge on siphon tube attached to top of receiver to give receiver pressure.
3. Thermometer on pipe leading from compressor to receiver.

4. Thermometer on water pipe leading to water space, to give temp. of cooling water.
5. Thermometer on pipe leading from water space to water tank to give temperature of water coming from machine.
6. Thermometer to give temp. of room from which the compressor drew its supply = dry bulb of Hygrometer.
7. Thermometer placed near the dry bulb one, with its bulb covered with moist waste = wet bulb of Hygrometer.
8. Tank and scales to give amount of cooling water used.
9. Barometer, to give atmospheric pressure.
10. Counter, attached to machine, worked from the cranks of indicator rig, to give the number of revolutions.
11. Two indicators, one attached to pipe leading to H.P. and the other to pipe leading to L.P. cylinders.

The tests were from 25 to 30 minutes duration, a longer time being impossible owing to the heating up of the bearings, as discussed above.

Four sets of cards were taken and readings of pressure and temperatures were taken every five minutes during each test. The readings of the counter and Barometer were taken between the taking of the cards and the weight of tank at the beginning and end of each test. The approximate time of each reading was noted to avoid mistakes, except in the case of the tank and counter which were accurate.

The pressure at the time of taking of each pair of cards was taken to enable one to obtain a mean pressure approximating that of the

whole test by increasing or decreasing the receiver pressure at time of taking of cards, since the opening of the indicator cocks was sufficient to produce a drop of from one half to one pound in the receiver, the receiver not being large enough to reduce this variation.

Calibration of Thermometers.

At the end of third test thermometers No. 4, 5, 6, 7 were placed in a bucket half full of water at about the temperature of the room and after some time were compared and the mean of the four taken as the correct temperature. The error of each was noted and their mean reading in each test corrected for this error.

thermometer No. 3 being at such a high temp. was assumed correct as we had no means at hand to calibrate it at these temperatures, under the conditions at which the test was made, and it was also just equal to the mean of the other four when placed in the bucket, as described above.

Calibration of Gauge.

The Gauge was calibrated by means of a Crosby Gauge Testing apparatus, the readings going up and coming down the scale being taken and their means as the uncorrected reading of gauge.

The difference between these means and the actual pressure due to the weights on the testing appa-

ratus gives the correction to be applied to the mean of the Gauge readings during each test; thus giving the corrected pressure.

The following are the readings obtained from the three tests:

Table-1- First Run. May-30-1899.

Time	Pressure	Thermometers. No 4	No. 5.	No. 3	Tank. Wt. Empty	Wt. Full.	Barometer	Hygrometer Wet Bulb	Dry Bulb	Counter
hrs. MIN.	lbs.									
12-7	92	82	100	213	706		29.975	72	85	84870
12-17	91.5	82	100	214				72	85	
12-22	91.5	81.5	101	214.5			29.977	71	85	
12-27	91	80	101	216				71	85	
12-32	92	80	102	216		769	29.972	71	85	87770
Sums	458	325.5	504	1073.5						Difference 2900
Means	91.6	81.1	100.8	214.7	Difference	63	29.975	71.5	8.5	Rev. per MIN. 116.
Corrections	-2.3	—	+.5	—				—	-.5	
Corrected Values.	89.3	81.1	101.3	214.7	Wt. per MIN.	2.52	29.975	71.5	84.5	

Table-I- Second Run. May-30-1899.

Time	Pressure	Thermometers No. 4	No. 5.	No. 3	Tank Wt. Empty	Wt. Full	Barometer	Hygrometer Wet Bulb	Dry Bulb	Counter
3-36	82.5	82	92	212	699.5		29.980	74	~~88~~	90688
3-41	82	~~83~~	97	215				74	~~88~~	
3-47	81.5	~~83~~	98	214			29.980	74	~~88~~	
3-52	84	~~84~~	101	213				74	88	
3-57	82	~~84~~	104	212				74	88	
4-01	82.5	~~84~~	104	213		76.1	29.980	74	88	93720
Sums	494.5	~~500~~	596	1279	Difference	61.5				Difference 3032
Means	82.4	~~83.3~~	99.3	213.2			29.980	74	88	Rev per Min. 121.3
Corrections	-2.0	~~+.5~~	+.5	—				—	-.5	
Corrected Values.	80.4	83.3	99.8	213.2	Wt. per Min	2.46	29.980	74	87.5	

Table I. Third Run. May - 30 - 1899.

Time	Pressure	Thermometers do. 4	do. 5	do. 3.	Tank. Wt. Empty	Wt. Full	Barometer	Hygrometer Wet Bulb	Dry Bulb	Counter
5-59	72.5	86	90	201	685		29.990	75	89	95815
6-04	71.5	87	93	206				75	89	
6-09	72	88	98	210				75	88	
6-14	72	88	99	212			29.980	75	88	
6-19	72.5	88	102	214				75	88	
6-24	72	88	104	215				74	88	
6-29	73.5	88	106	218		760.5	29.995	74	88	99508
Sums	506	613	692	1476	Difference	75.5		523	618	Difference 3693
Means	72.3	87.6	98.9	210.9			29.992	74.7	88.3	Rev per Min. 123.1
Corrections	-2.7	—	+.5	—					+.5	
Corrected Values	69.6	87.6	99.4	210.9	Wt. per Min	2.51	29.992	74.7	87.8	

Calibration of Bourbon Gauge No. 210018.

Table -2-

Up. Scale	Down Scale	Means	Corrections.
67.2	67.8	67.5	- 2.5
72.1	73.3	72.7	- 2.7
76.9	77.9	77.4	- 2.4
81.2	83.2	82.2	- 2.2
87.4	87.3	87.4	- 2.4
92.4	92.0	92.2	- 2.2
97.1	97.3	97.2	- 2.2

Calibration of Indicator Springs.

The indicator springs were calibrated by the method of finding the mean spring scale, the Crosby Testing apparatus being also used for their calibration since the conditions under which the indicators were used was more nearly that of the Gauge Testing Apparatus than the usual method of testing by means of steam and a steam gauge.

The means of the distances on the indicator cards taken at each ten lbs. added in compressing the spring, or subtracted in expanding the spring were summed up and the mean distance taken in the case of the 60 lb. spring.

This then divided into the actual weight to which this distance is

due, i.e., ten lbs, gave the mean spring scale as 60.5 lbs.

The same method was employed in calibrating the 20 lb. spring except that only five lbs. were added and subtracted instead of ten.

Table-3-

Calibration of Low Pressure Spring.

lbs.	Inches going up.	Inches going down	Means	Scale of Spring.
0 - 5	.23	.23	.23	
5 - 10	.24	.27	.255	
10 - 15	.24	.26	.25	
15 - 20	.25	.25	.25	
20 - 25	.24	.23	.235	21.05
25 - 30	.23	.23	.23	
30 - 35	.25	.21	.23	
35 - 40	.24	.20	.22	
Sums			1.900	
Means.			.2375	

Table -4-

Calibration of High Pressure Spring

Lbs.	Inches up	Inches Down	Means	Scale of Spring
10-20	.16	.17		
20-30	.17	.15	.165	
30-40	.17	.16	.16	
40-50	.17	.16	.165	
50-60	.16	.17	.165	
60-70	.16	.17	.165	60.5
70-80	.16	.17	.165	
80-90	.16	.18	.17	
90-100	.18	.16	.17	
Sums			1490	
Means			.165.5	

From the four sets or pairs of cards of each test mean cards were now drawn, corrected for the incorrect indicator motion. The method used was as follows;

The indicators were attached to the machine and marks drawn upon one card corresponding to the position of tenths of the stroke of the corresponding piston as determined by a steel rule. This was repeated with the other indicator attached to the other cylinder for tenths of its piston's stroke, in each case making a complete revolution. Starting with the piston of the high pressure cylinder at the highest point of its stroke, the points obtained for the downward part of the stroke were laid off on one

actual card (see Fig 1) and ordinates erected on the four cards of one test.

Actual Cards — Second Run.
Low Pressure Card.

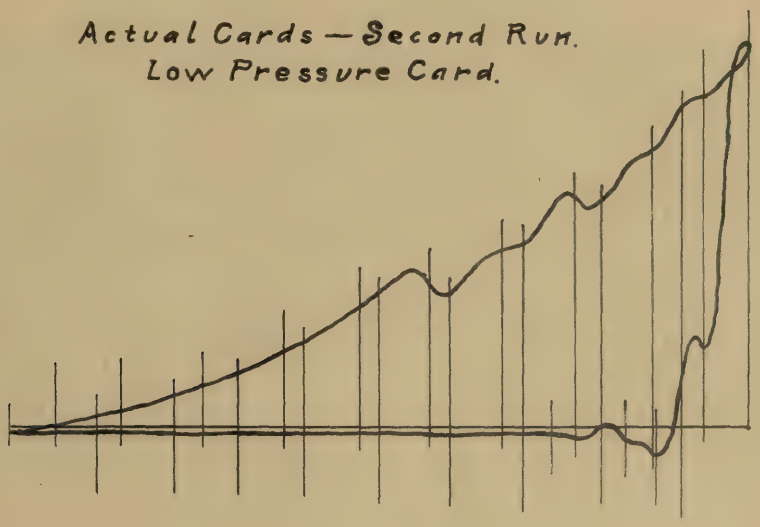


Fig. 2.

Actual Cards — Second Run.
High Pressure Card.

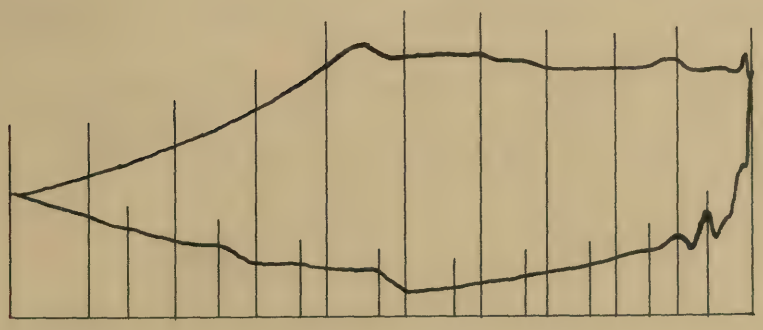


Fig. 1.

The mean card was now drawn by taking the mean length of the four actual cards and dividing it into ten parts and erecting ordinates. The mean of the heights taken on the ordinates of the four cards were laid off on corresponding ordinates of the mean cards starting from the correct end of the stroke, or in this case when its piston starts to compress on its downward stroke.

The ordinates of the other half of the revolution were erected on the four cards and the mean heights erected on the corresponding ordinates of the mean card.

This method was repeated with the low pressure (L. P.) cylinder except that since the pressure

is almost constant except at the admission end of the cards only two of the ordinates are shown on the actual card Fig 2.

Note.

The order of the runs is reversed in the thesis from that actually used, as will be seen by comparing Table I with the mean cards, Figs. 3, 4, and 5.

MEAN CARDS - FIRST RUN

HIGH PRESSURE

AREA-3.13 LENGTH-3.87

LOW PRESSURE

AREA-3.07 LENGTH-3.84

Fig. 3.

UNIVERSITY of PENNSYLVANIA LIBRARIES

MEAN CARDS - SECOND RUN

HIGH PRESSURE.

AREA - 3.24 LENGTH - 3.86

LOW PRESSURE.

AREA - 3.04 LENGTH - 3.84

Fig. 4.

MEAN CARDS - THIRD RUN

HIGH PRESSURE

AREA - 3.55 LENGTH - 3.88

LOW PRESSURE

AREA - 2.94 LENGTH - 3.85

Fig. 5

Efficiency of Compression.

After drawing the mean cards for the three tests the second run set were used to obtain the efficiency of compression.

By efficiency of compression is generally meant the ratio of the work done in compressing to a certain pressure along the isothermal to that actually done by the machine, since it has been the aim of the makers of air compressors to approximate as nearly as possible to isothermal compression.

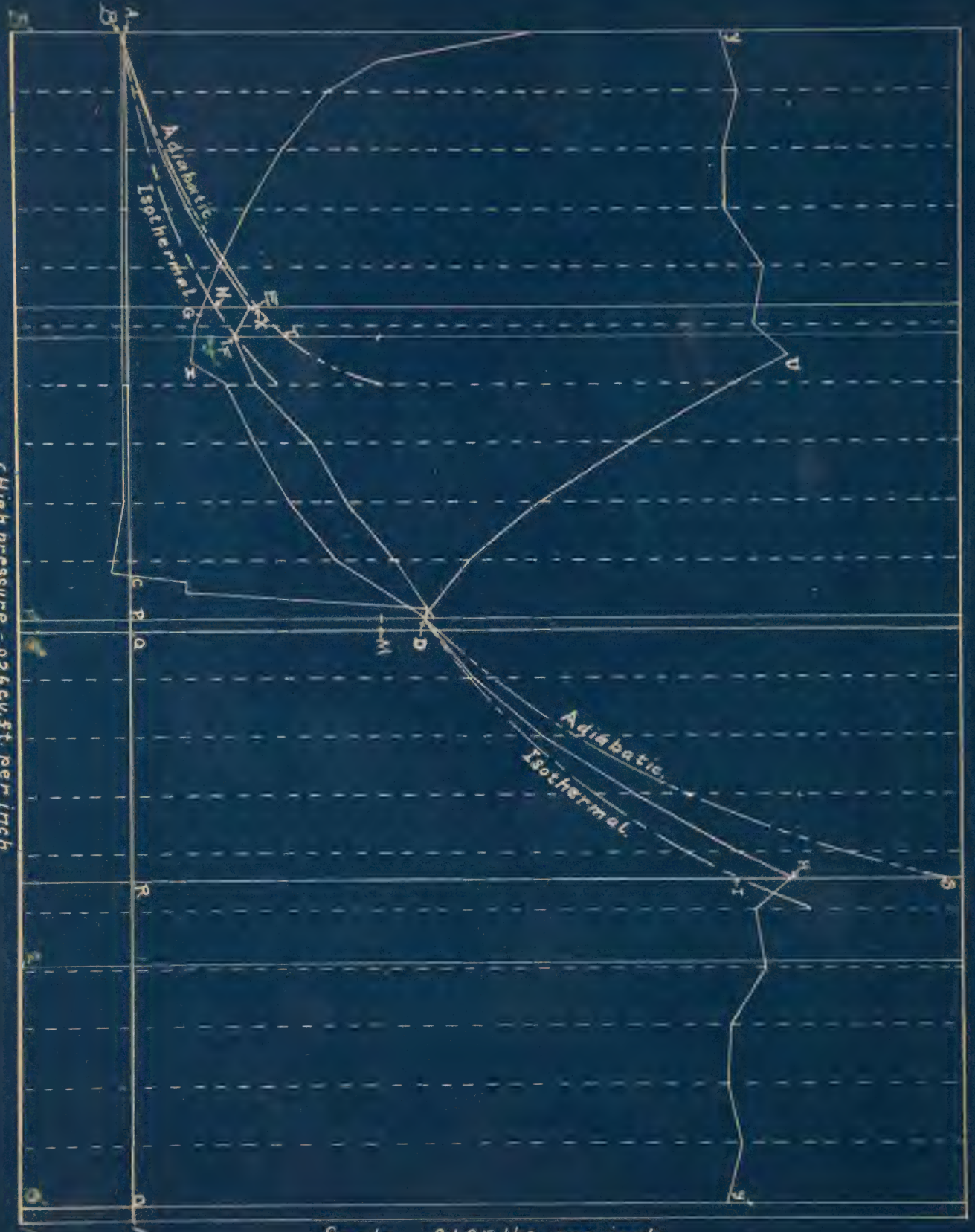
The reason for this is that although the greatest amount of heat would be usefully employed if the compression were adiabatic, a portion of this heat would be wasted

if the air were not immediately used, since it would cool off in the receiver or reservoir.

If now the compression is made at a constant temperature, or along the isothermal no heat would be wasted, since the receiver would be at the same temperature as the outside air, although less work would be done on the air.

To determine the efficiency we found the heat lost on the different parts of the compression line due to the non-isothermal compression. To do this we took the second run mean cards and reduced them to the same pressure scale, that of the low pressure card. See Diagram I

HEAT CARDS — DIAGRAM - 1.


Adiabatic
Isothermal
Adiabatic.
Isothermal.
Scales { High pressure - .026 cu. ft. per inch
Low pressure - .086 cu. ft. per inch.
Scale - 21.05 lbs. per inch.

AA' is the atmospheric line and YUDZ the reduced H.P. card. For clearness the compression part of the H.P. card is reproduced at DH. From the point B, the beginning of the compression, the isothermal BN is drawn using BB' as the pressure, and the volume that equal to the volume of the L.P. cylinder plus the L.P. clearance volume as given on page 4 reduced to the correct scale or AP, plus PQ, which equals AQ. Q' is therefor the point of zero pressure and zero volume from which the isothermal BN is drawn.
The adiabatic BXT was then drawn for comparison with the actual line BE obtained from the mean L.P. card BEFDC.

Knowing the pressure BB' and the volume AQ we found the constant of $PV^{1.41}$ = constant and then by assuming different volumes we found the corresponding pressure.

These methods were repeated to obtain the isothermal DI and the adiabatic DS using PQ' as the pressure and the volume = volume of H.P. cylinder + H.P. clearance (see p 5) = PO + OA' = PA'. A" is therefore the point from which the isothermal line is drawn.

To obtain the loss on the L.P. compression line we must find the temperature at E since the loss will be equal to the difference of the heat at beginning of compression and at E or $C_v (T_E - T_B)$ since the volume is constant.

We know the temperature at N is equal to that at B since these points are both on the isothermal.

Therefor the loss = $C_V (T_E - T_N)$. But

$T_E = \frac{P_E V_E}{MR}$ and $T_N = \frac{P_N V_N}{MR}$

Assuming no loss due to leakage the mass M will be constant; R is constant; $V_N = V_E$ and therefor the temperature will vary directly as the pressure or $\frac{T_E}{T_N} = \frac{P_E}{P_N}$, absolute pressures being used. Since the pressures **are** proportional to the ordinates the ratio of the heights alone is used to obtain T_E where $T_E = \frac{P_E}{P_N} T_N$, T_N being the mean corrected temperature of the dry bulb on thermometer No. 6. (see table I).

Note.

All thermometers register Fahrenheit degrees.

Heat Loss on L. P. compression Line.

$P_E = 1.54''$ $P_N = 1.31$ $T_N = 548.2°$

$\therefore T_E = \frac{1.54}{1.31} \times 548.2 = 645.5°$

C_v for air $= .1691$

Heat loss $= C_v (T_E - T_N)$

$= .1691 \times (645.5 - 548.2)$

$= .1691 \times 97.3 = 16.45$ B.T.U. per lb. air.

The H.P. loss was calculated by the same method except we now know the temperature at the point H on the actual compression line, equivalent to the temperature of compressed air, the mean reading of thermometer No. 3., and must find the temperature at I on the isothermal.

Heat Loss on H.P. compression line

$P_H = 5.09''$ $P_I = 4.75''$ $T_H = 673.9°$

$\therefore T_I = \frac{4.75}{5.09} \times 673.9 = 629°$

$\therefore$ Heat Loss $= C_v (T_H - T_I)$

$= .1691 (673.9 - 629.0)$

$= .1691 \times 44.9 = 7.59$ B.T.U. per lb. of air.

To find the loss in the interval between the L.P. compression line and H.P. compression line, or from E to D we know only the temperature at D which equals that at I, since it is on the same isothermal. But we may consider the compression as starting from the point F, since at E the valves open from the L.P. cylinder into the intermediate clearance (I.C.) and the air expands owing to the large clearance volume = 91.8 cu. in., nearly

one half the volume of the H.P. cylinder = 173.2 cu. in. at F this expansion is complete and the air partly in the L.P. cylinder and clearance, in the I.C., and partly in the H.P. cylinder and clearance is compressed on some line between those shown at FD and ZD.

To obtain the pressure at the point of starting of the isothermal we took a mean value between that of the L.P. ordinate through F and a corresponding ordinate in the stroke of the H.P. through G; the cards as shown not being corrected for the angularity of the connecting rod.

This correction will be explained in drawing the combined cards.

The mean of the ordinates gave the point F' from which the isothermal could be drawn cutting the ordinate

PD at M., which we wanted to obtain.
The clearance volume used in drawing the isothermal line = H.P. clearance plus L.P clearance plus I.C. = P1, or 2 is the point of zero volume and zero pressure.

Knowing the temperature at D by the method explained above we can obtain the temperature at M.

Heat Loss on Intermediate Line

$P_M = 2\frac{24}{64}$" $P_D = 2\frac{42}{64}$" $T_D = 197° + 460.7 = 657.7$

$\therefore T_M = \frac{152}{170} \times 657.7 = 589°$

Heat loss = .1691 (657.7 − 589)

= .1691 × 68.7 = 11.63 B.T.U per lb. of air

$\therefore$ Total loss = 16.45 + 7.59 + 11.63 = 35.67 B.T.U. per lb. of air.

To find the total heat loss per

minute we must find the amount of air used.

The amount of air will depend on the pressure at entrance and the volume used per minute.

To find the pressure at entrance we will first find the atmospheric pressure given by the barometer, see Table I.

Barometric pressure =

$$\frac{29.98 \times 13.6 \times .434}{12} = 14.74 \text{ lb.}$$

per sq. in.

This pressure is equal to the pressure of the air plus that due to the vapor tension of the moisture contained in the air. To obtain the correct vapor pressure we must find the relative humidity.

Table I gives the mean values of

the wet and dry bulbs.
a table in Kent's Mechanical Engineer's Pocket Book, page 483, gives by interpolation 47.6 % humidity on a basis of saturation = 100 %.

Mean dry bulb = 87.5
Mean wet bulb = 74.0
Difference = 13.5

at 82° humidity = 46.5 %
at 92° humidity = 48.5 %
∴ 10° = 2.0 %
and 5.5° = 1.1 %

To reduce to a dry air basis the humidity at 87.5° = 46.5 + 1.1 = 47.6 %.

The Pressure of water vapor given in Peabody's Steam Tables gives for
88° = .6502
87° = .6299 from which 87.5° = .6400

Therefor the corrected vapor pressure
$= .476 \times .64 = .305$ lbs. per sq. in.
and the corrected air pressure is
$14.74 - .305 = 14.43$ lbs. per sq. in.

The mass per revolution is given by the formula $PV = MRT$ by combination of Boyle's and Charles' Laws.
Or $M = \frac{PV}{RT}$ where P = pressure given above $= 14.43 \times 144$ lbs. per sq. ft.
R is a constant $= 53.37$
T = absolute temperature of dry bulb
$= T_B = 87.5 + 460.7$ and V is the volume in cu. ft. to point of opening of valves, as discussed above.

$$V = \frac{3.58''}{3.85''} \times 570.18 \text{ cu. in} = 531 \text{ cu. in.}$$

$$\therefore M = \frac{14.43 \times 144 \times \frac{531}{1728}}{53.37 \times (460.7 + 87.5)} = .02175 \text{ lbs.}$$

Mass per minute = M multiplied by the number of revolutions per minute from Table I =

.02175 x 121.3 = 2.638 pounds of dry air.

And the total heat loss per minute =

2.638 x 35.67 (see page 40)

= 94.15 B.T.U.

The mass of air in 531 cu. in =

.02175 lb. (see page 43)

Therefore the mass in 1 cu. ft =

$\frac{1728}{531}$ x .02175 = .0709 lbs.

Indicated Work.

The work done on the air by the machine is given by the mean corrected cards for this run.

Now, Horse Power (HP) = $\frac{PLAN}{33000}$ where P is the mean effective pressure (M.E.P), L the length of the stroke in feet, A the area of the piston in sq. in., and N is the number of strokes per minute = number of revolutions in single stroke engines.

Therefor engine constant = $\frac{LA}{33000}$ = $\frac{.5 \times 95.03}{33000}$ = .00144 for L.P cylinder, the length of stroke being 6 inches and diameter 11 inches.

The M.E.P = $\frac{3.04}{3.84} \times 21.05$ = 16.68 pounds per sq. in. in which 3.04 = area of card (see Fig. 4) in sq. in, 3.84 = length in inches of same card and 21.05 = corrected spring scale. (see Table III)

Therefor HP = .00144 x 16.68 x 121.3 = 2.91

The engine constant for H.P. cylinder is likewise = $\frac{L\,A}{33000}$ = .000429 in which L is the same as befor and A = 28.3 sq. in, or $\frac{.5 \times 28.3}{33000}$ = .000429

And M.E.P. = $\frac{3.24}{3.86} \times 60.5$ = 50.8 pounds per sq. in. 3.24 being the area, 3.86 the length and 60.5 the corrected spring pressure of the mean H.P. card (see Fig. 4, and Table 4.).

The H.P. is then = .000429 x 50.8 x 121.3

= 2.64

And the total H.P. = 2.91 + 2.64 = 5.55

This total H.P. is equal to

$\frac{5.55 \times 33000}{778}$ = 235.5 B.T.U. per minute.

From page 44 we have 94.15 B.T.U loss per min. and therefore work of isothermal compression = 235.5 - 94.15 = 141.35 B.T.U. and therefore the efficiency = $\frac{141.35}{235.5}$ = 60.0 %

For commercial use we will find the HP necessary to compress 1000 cu. ft of dry air to the mean pressure of the run. From page 44 the mass per cu. ft. of dry air = .0709 lbs. Therefore the number of cu. ft. compressed assuming no loss = total mass per minute divided by the mass of dry air per cu. ft. Thus = $\frac{2.638}{.0709}$ = 37.20 cu. ft.

HP per min. per 1000 cu. ft. = $\frac{1000}{37.2} \times 5.55 = 149.0$

Owing to want of time we were unable to find the efficiency of compression of the other two runs., but found the HP per 1000 cu. ft. The method is the same as for the second run, and the relative humidity being nearly the same and therefor the pressure of the air at entrance was assumed

constant the error so produced being less than one half of one per cent

First Run.

To point of opening of valve = 3.61".

Therefore volume =

$$\frac{3.61}{3.84} \times 570.18 = 536.0 \text{ cu. in.}$$

and mass per revolution is equal to

$$M = \frac{14.43 \times 144 \times 536}{53.37 \times 1728 \times 548.5} = .02195 \text{ lbs.}$$

For comparison with the other tests the number of revolutions in the first and third tests will be assumed that of the second test = 121.3 per minute, in order

to have only two variables,
namely, H.P and pressure.

The weight per minute =

$.02195 \times 121.3 = 2.66$ lbs.

The mass per cu. ft (see method p 44)

$= \frac{1728}{536} \times .02195 = .0708$ lbs.

Therefore the number of cu. ft. compressed
per minute $= \frac{2.66}{.0708} = 37.6$

H.P from cards, Fig. 3.

L. P. cylinder. M. E. P. $= \frac{3.07}{3.84} \times 21.05 = 16.83$

$H.P = .00144 \times 16.83 \times 121.3 = 2.94$

H. P. cylinder. M. E. P. $= \frac{3.13}{3.87} \times 60.5 = 49.0$

$H.P = .000429 \times 49 \times 121.3 = 2.55$

Total $H.P = 2.94 + 2.55 = 5.49$

H.P per 1000 cu. ft per min. $= \frac{1000}{37.6} \times 5.49 = 146.6$

Third Run

To point of opening of valves = 3.56".

The volume = $\frac{3.56}{3.85} \times 570.18 = 527$ cu. in

The Mass per revolution =

$$\frac{14.43 \times 144 \times 527}{53.37 \times 1728 \times 545.2} = .02175 \text{ lbs.}$$

Mass per minute =

$.02175 \times 121.3 = 2.635$ lbs.

The mass per cu. ft. (see method p 44) =

$$\frac{1728}{527} \times .02175 = .0713 \text{ lbs.}$$

Therefore the number of cu. ft. of dry air compressed per minute =

$$\frac{2.635}{.0713} = 37.0 \text{ cu. ft.}$$

Horse Power from cards Fig. 5.

L. P. cylinder, $M.E.P. = \frac{2.94}{3.85} \times 21.05 = 15.7$

$H.P. = .00144 \times 15.7 \times 121.3 = 2.74$

H. P. cylinder, $M.E.P. = \frac{3.55}{3.85} \times 60.5 = 55.8$

$H.P. = .000429 \times 55.8 \times 121.3 = 2.90$

Total $H.P. = 2.74 + 2.90 = 5.64$

$H.P.$ per min. per 1000 cu. ft. $= \frac{1000}{37.0} \times 5.64 = 152.3$

To enable a comparison to be made between the three runs the amount of cooling water was kept approximately constant as shown in Table I. This we were enabled to do since the tests were all run in one day and after first regulating the amount by means of a valve it was allowed to run at this rate.

Volumetric efficiency.

By this is meant the ratio of the amount of air actually compressed per stroke to the L. P. piston displacement.

The volume actually compressed as explained above page is that from the time the valve opens expressed in cu. inches

For 70 lbs: $\frac{3.61}{3.84} \times 570.18$ cu. in. = 536 cu. in.

Therefor volumetric efficiency

$= \frac{536}{570.18} = 94\%$

For 80 lbs: $\frac{3.58}{3.85} \times 570.18 = 531$ cu. in

volumetric efficiency

$= \frac{531}{570.18} = 93.1\%$

For 90 lbs: $\frac{3.56}{3.85} \times 570.18 = 527$ cu. in.

volumetric efficiency

$= \frac{527}{570.18} = 92.4\%$.

The Combined Cards.

For the purpose of making comparisons the combined cards, Diagram 2, from the second run were made.

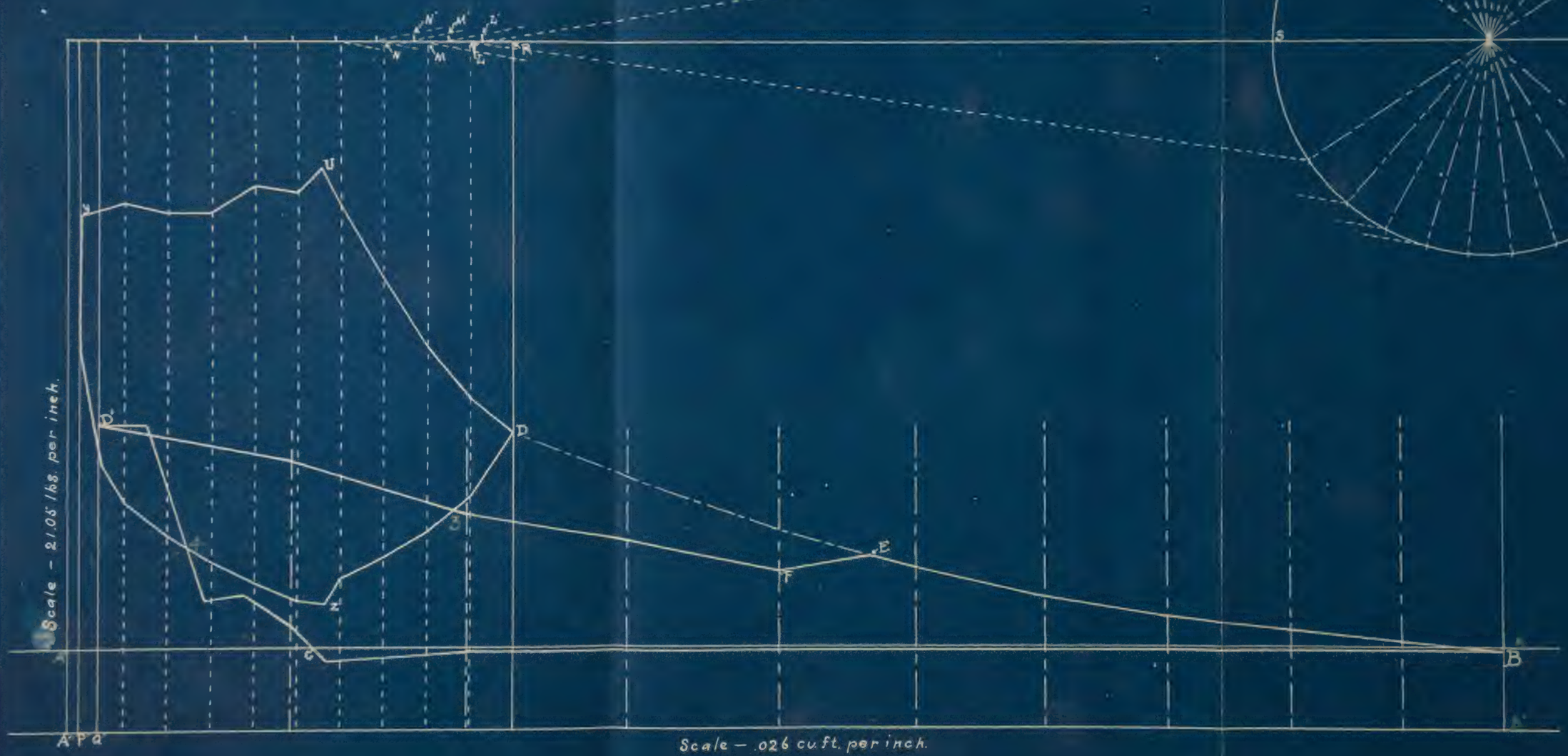
The method used was as follows:

The H.P. piston was moved down and up one tenth of its stroke at a time, and marks drawn on a card of the L.P. indicator. These points were laid off on the L.P. cards of this run and ordinates erected. This corrects for the angularity of the strings as well as the angularity of the connecting rod.

Then (Diagram 2) the positions of the L.P. piston were found for tenths of the stroke of the H.P. piston, the distance R.S. being the length of the connecting rod, reduced to the same scale as S.T., the length of the card representing

the length of the stroke.
Therefor R.L, L M, M N, etc on the H.P. card = R L', L' M', M' N', etc on the L.P. stroke. To the right scale these lengths were laid off on line A A' and ordinates erected.
These ordinates then actually correspond to those of the H.P. stroke and the heights taken from the actual cards gave the lengths of the ordinates through which the L. P. card - D'E B C - reduced card was drawn, the lettering being the same as in Diagram I.
If drawn correctly the area D', 3, Z, 4, will equal appproximately the area DEF 3, and this is seen to be nearly so.

COMBINED CARDS — DIAGRAM - 2.



Conclusions.

From Diagram I it is seen that the actual compression L. P. line approximates the adiabatic while the H. P. compression line is better, or nearer the isothermal.

The first is due to the inadequate supply of water, the volume of air being so large that the cooling surface is small in comparison.

This result is shown by the large heat loss of 16.45 B. T. U. per pound of air.

The water flows around both the H. P. and L. P. cylinders by one common passage, and is heated up simultaneously, that is, the water is not heated by one cylinder before it goes to the other.

The smaller loss in the H. P.

cylinder is therefore due to the greater ratio of cooling surface to the volume on which it acts, since the surface varies as the diameter, while the volume varies as the diameter squared.

The comparatively large loss in the intermediate clearance, since its volume is about one half that of the H.P. cylinder, is probably due to the method of cooling, only the part of the top and sides being cooling surfaces, while the friction of the air in passing the valves is also counted in this part by the methods used to obtain the loss in this place.

The efficiency of compression is probably higher than it ought to be

because it was assumed that all the air entering was compressed, that is, that there was no loss due to leakage.

An effort was made to determine this leakage by means of the formula $M = \frac{(P-p)V}{RT}$. P being the total pressure and p the vapor pressure.

This would give the mass at the highest point of compression if either the correct pressure of the water vapor or that of the air could be found, since the combined pressure is known and the volume and temperature. But since the vapor pressure depends on the degree of saturation, and this was unknown, it was impossible to find the pressure.

Assuming the air saturated, which is impossible, at the temp-

erature of the compressed air, an approximate solution can be obtained, since the vapor tension would then be 14.7 pounds, corresponding to the temperature of the air.

Taking as an example the second run:

Total pressure = 87.12 pounds.

vapor tension = 14.7 pounds

The difference = 72.12 pounds per sq. in.

The volume = volume of air at point H. (Diagram I) = .0572 cu. ft.

Temperature = 213.3° = 674° absolute from Table I

Therefore $M = \frac{72.12 \times 144 \times .0572}{53.37 \times 674} = .02088$ lbs.

Mass entering = .02175 lbs.

Loss = .02175 - .02088 = .00067 lb.

$\frac{.00067}{.02175} = .031$ or 3.1% loss.

This is only a rough approximation and is, therefore of no practical value except in showing that there probably is a loss.

It will be noticed in the Figs. 4, 5, and 6 that the area of the L.P. card decreases as the pressure in the receiver increases. This is because the valves open later into the L.P. on the return stroke, as shown by the volumetric efficiency also decreasing in about the same ratio. (see page 53).

The HP per 1000 cu. ft. per minute increases with the pressure to which the air is compressed, as would be expected.

The rate of increase, however, is greater for the increase from 80 to 90 lbs. pressure, than from 70 to 80 lbs. pressure, the exact varia-

tion in the rate being beyond the scope of this thesis, as only three points were determined.

University of Pennsylvania Library
Circulation Department

Please return this book as soon as you have finished with it. In order to avoid a fine it must be returned by the latest date stamped below.

MAY 10 1983

W

M-719

3 1198 03073 2957
N/1198/03073/2957X

Towne
378.748
POS 1899.8

Biddle
Analysis of a Compound
Belt Driven Air
Compressor

DATE	ISSUED TO

TOWNE
378.748
POS 1899.8

3 1198 03073 2957
N/1198/03073/2957X